8th International Tribology Conference
Osma internacionalna konferencija o tribologiji
Beograd, 8. - 10. oktobra 2003.

ANY THEORETICAL AND EXPERIMENTAL ANALOGIES ON TRIBOLOGICAL RESEARCH OF NON-CONTACT SEAL DEVICES

Falticeanu CONSTANTIN, Manea MITICA, Ciortan SORIN,
"DUNAREA DE JOS" University of Galati, Romania

S u m m a r y

The paper presents some results of the theoretical and experimental studies concerning the tribological behaviour and the effectiveness of non-contact seal devices with axial-cylindrical clearances. The tribological behaviour of seals was determined by studying one of the effects of friction in the clearances, exactly of the thermal regime of the seal devices, and the effectiveness was studied by determining the leakage flow-rates, respectively the ways of diminishing them. There are presented calculus relations for the determinative temperature of the sealed liquid inside the clearances and for the leakage flow-rate in the seals.

Keywords: *non-contact seal device, leakage flow-rate, determinative temperature, axial-cylindrical clearance, tribological behaviour, effectiveness.*

1. INTRODUCTION

Non-contact seal devices with axial-cylindrical clearances, figure 1,[11] eliminate the contact between the sealed surfaces and have for effect the avoidance of all disadvantages, associated with friction, deterioration and lubrication, that can be found at the contact mobile seals. There are many situations where, in the context of the improvement of the fiability and performances of technical systems, the use of non-contact seal devices is a major and at the same time unique solution.

The lack of contact between the sealed surfaces leads, obviously, towards the appearance of a clearance through which liquid passes at a certain flow-rate and inside which friction phenomenon take place. The friction inside the non-contact seals' clearances is a complex phenomenon that has for consequences the thermal process with energy loss (heat) as well as the deterioration process of the active surfaces of the clearance [1,3,4,6,7,11].

The study of the tribological behaviour of non-contact seals imposes a research in which to take in consideration at least the following aspects: the development, evolution and consequences of the thermal regime, respectively, the appearance, evolution and effects of deterioration [6,11].

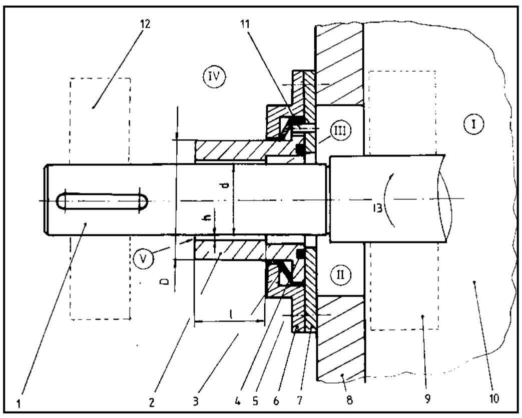

Figure 1 – Geometrical and functional model of non-contact seal device with axial-cylindrical clearance.

1 – seal shaft; 2 – floating socket; 3 – discus spring; 4 – "O" ring seal; 5 – screws; 6 – lid; 7 – basis plate; 8 – sealed technical system carcass wall; 9 – shaft rotation device; 10 – sealed environment; 11 – centring and blocking bolts; 12 – element through which the shaft transmits the movement; *d* – seal nominal diameter; *h* – clearance width; *l* – clearance length; *D* – floating socket exterior diameter.

The effectiveness of non-contact seals can be expressed by the size of the leakage flow-rates and by the ways of their diminution, up to the point of their annulment. When the annulment of the leakage is not possible, it is necessary to predict the value of the flow-rates as accurately as possible together with the factors that influence it [3,6,7,11,12].

The present paper consists in a theoretical and experimental study of some tribological and effectiveness aspects of non-contact seal devices with axial-cylindrical clearances, destined to liquid environments.

2. THE USED METHODOLOGY OF RESEARCH AND MATERIAL BASE

The base of the theoretical studies consists in the following laws and fundamental equations: the equations for energy, mass and impulse conservation, the internal energy equation, the differential equations for the liquid particles' trajectories, the laws of *Newton* and *Fourier* (concerning the heat flow, respectively, the thermal convection), etc.[6,10,11].

The study of the liquid flow through the clearances has aimed at determining the flowing regime (using the critical flow-rates and the trajectory methods) [6,11], the liquid particles' flowing velocities and the liquid pressure distribution inside the clearances [1,6,11,12]. That is why the condition for total avoidance of direct contact between the shaft and the bore was established and there were taken in consideration the following: the sealed liquid parameters modification due to temperature, the clearance width modification due to thermal dilatations and elastic deformations produced by the liquid pressure and the liquid pressure modification due to the clearance geometry at the liquid's entrance, respectively exit.

The tribological behaviour research of the non-contact seal devices was studied only through the analysis of a fluid friction effect inside the clearance exactly of their thermal regime.

The determination of the power loss through friction was made on condition that any direct contact between the clearance surfaces is missing and it was considered that it transforms entirely into heat, that dissipates uniformly in the entire liquid mass inside the clearance.

It was considered that inside the clearances the sealed liquid temperature value doesn't change in time, that the isothermal temperatures are fixed in space and that the temperature field is permanent or stationary.

The existence of the temperature field in the liquid mass inside the clearance made necessary the determination of an average value of the temperature, the *determinative temperature*, that intervenes in the leakage calculus and for which the value of the other physical parameters of the sealed liquid must be adopted [2].

The experimental studies and researches were made using an experimental stall, figure 2, [6,11] which was executed in a modular construction, the four modular components (the experimental module, the hydraulic module, the shaft rotation module and the measurement equipment module) forming a constructively and functionally integrated ensemble, fulfilling all the requirements and performance criteria imposed by the subject and objectives of the research.

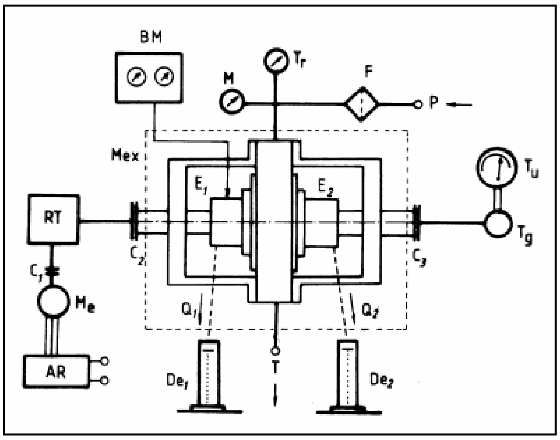

Figure 2 – Principle scheme of the experimental stall

F – fine secondary filter; **Tr** – temperature translating device; **M** – manometer; **Tg** – tachometer; **Tu** – electromagnetic revolution translating device; **BM** – measurement bloc; **Me** – electrical motor; **AR** – alimentation-adjustment bloc; **RT** – revolution regulator; **C1, C2** – couplings; **De1, De2** – flow-rate measurement tool; **E1, E2** – the experimentally studied seals; **Mex** – experimental model.

For performing the experimental studies the M30 mineral oil was used as the sealed liquid. The connection between the hydraulic module and the experimental model is made in points *P* (of the pump) and *T* (towards the tank).

The experimental researches were made for non-contact seal devices with simple axial-cylindrical clearances, with fixed or mobile socket.

During the experimental studies the leakage flow-rates have been determined in the form of volume flow-rates, expressed in m^3/s; they were determined only after the seals reached the thermal equilibrium and after the seal functional parameters' stabilization was noted.

The actual measurement of the flow-rates consisted in determining, with the help of graded receptacles, the volumes of the leakage in a certain interval of time whose value was established through timing (the intervals of time were established between 10 and 30 minutes).

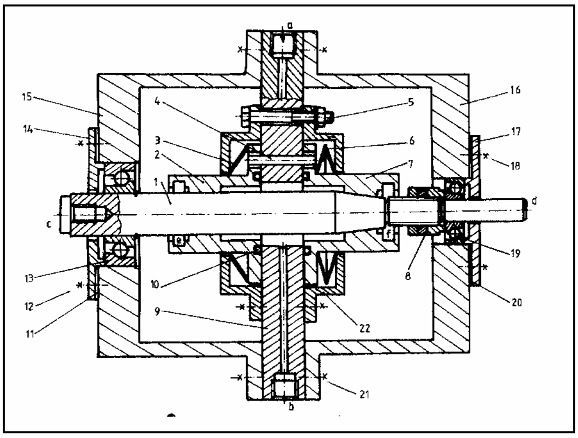

Figure 3 – Experimental model for simultaneous research of two non-contact floating socket seal devices

1 – seals' shaft; 2 – axial-cylindrical socket; 7 – axial-conical seal socket; **a** – the entrance of the pressured liquid inside the sealed device; **b** – sealed liquid exit from the sealed device; **c** – seal shaft rotation device area; **d** – coupling area of revolution translating device; **e** – exit from the axial seal; **f** – exit from the conical seal.

For all effectuated determinations the external environment of the studied seal devices was air, taken in consideration at the atmospheric pressure p_0 = 10^5Pa and having a normal circulation around the seals. The sealed liquid pressures, that were determined in the form of relative pressures, were measured using the manometers with elastic tube having the class of precision 0,6 and the measurement precision 0,1 x 10^4 Pa. For measuring the pressure in the axial clearances' area special pressure measurement points were set disposed on cylindrical helixes lengthways the seal.

The measurement of the liquid temperature inside the tank as well as the measurement of the seal device surrounding environment temperature was made with glass thermometers with liquid, having the measurement precision of 0,5 ^{0}C. The sealed liquid temperature, at it's entry inside the device's body was determined using a bimetal thermometer. The external surfaces of the socket and of the seal disks were determined using fast surface thermocouples, with thermal inertia under 3s and measurement precision of 0,5^{0}C. The determination of the liquid temperature inside the seal clearances was made, indirectly, by determining the temperatures inside the seal socket wall with fast thermocouples, ø = 1,5 mm, under the circumstances of an almost complete lack of air currents around the seal.

3. SOME OBTAINED RESULTS (axial-cylindrical seal devices)

The hypotheses in witch the theoretical studies have been made are: the sealed liquids are Newtonian; inside the clearance the liquid motion is laminary and permanent; the sealed liquid perfectly adheres to the solid surfaces of the clearance which it fills completely; the width of the clearances is much smaller compared to the nominal diameter and the seal length; we can therefore neglect the mass forces compared to the viscosity ones; the very small width of the clearances allows considering the liquid velocity component on the *Oy* axis equal to 0; the seal shaft execution tolerances and the socket execution tolerances, the montage and the functioning of the seal device don't lead to the changing of the clearance width lengthways the seal or radially; the seal shaft rotates, constantly in time, with the angular speed ϖ ; the seal socket is considered to be fixed and concentric with the shaft.

The flowing regime of the sealed liquids inside the clearances is laminary because the actual leakage flow-rates are much smaller than the critical flow-rates, and the liquid particles trajectories are regulate parallel curves (cylindrical helixes with an approximate constant step) [6,8,11].

By integrating the differential equations of the liquid pressure inside the clearance.

$$\frac{d}{dx}\left(\frac{h^3}{\eta}\cdot\frac{dp}{dx}\right)=0 \qquad (1)$$

where: $h = h(x)$ is the local width of the clearance, $p = p(x)$ is the local liquid pressure, η, the dynamic viscosity of the liquid and x the axial co-ordinate of the clearance, the local pressure distribution equation for the sealed liquid inside the clearance could be found,

$$p(x) = 1/a \cdot \{[((\delta + a\cdot p_2)^4 - (\delta + a\cdot p_1)^4)\cdot x / l + (\delta + a\cdot p_1)^4]^{0,25} - \delta\} \qquad (2)$$

where: $\delta = h_0 + \Delta h_T$; $a = \Delta h_P / p(x)$; $h(x) = h_0 + \Delta h_T + \Delta h_P$; h_0 is the clearance width at 20^{0}C; Δh_T is the local width growth due to thermal dilatations; Δh_P is the clearance local elastic deformation due to the sealed liquid pressure; p_1 and p_2 are the sealed liquid pressure when entering, respectively, exiting the clearance (for p_1 and p_2 specific calculus relations were determined taking in consideration the clearance geometry at the liquid's entering, respectively exit [6,9,11], the sealed liquid density and the leakage flow-rate value); l is the clearance length.

Relation (2) renders the fact that the liquid pressure distribution inside the axial-cylindrical clearance, in the conditions of the clearance length modification due to temperature and sealed liquid pressure, is non-linear.

For the power loss through friction, in the liquid volume inside the axial-cylindrical clearance, the calculus relation was determined, similar to the *Petroff* relation [10]:

$$P_f = \pi \cdot \eta \cdot \omega^2 \cdot d^3 \cdot l \,/\, 4h \qquad (3)$$

where d is the nominal seal shaft diameter.

Considering that: the power loss through friction transforms integrally into heat, and the heat evacuation from inside the clearance to the exterior of the seal is made integrally through it's body, inside the clearance the thermal regime is stationary and the liquid temperature is constant in comparison with the *oz* axis and the unitary thermal heat flow inside the clearance and the sealed liquid physical properties are constant with temperature (inside the clearance), by integrating the energy conservation equation the equation for the sealed liquid temperature field inside the axial-cylindrical clearance was determined. On the basis of the temperature field equation and defining for a transversal clearance section the sealed liquid determinative temperature (T_f) using relation (4) the following calculus expressions were found:

$$T_f = (T_a + T_P)/2 \qquad (4)$$

$$T_f = \pi \cdot \eta \cdot \omega^2 \cdot d^3 \cdot l \,/\, 4 \cdot h \cdot K + T_0 \qquad (5)$$

$$T_a = \eta \cdot \omega^2 \cdot d^3/4 \cdot [\pi \cdot l/h \cdot K + 1/4 \cdot \lambda_f (d+2h)] + T_0 \qquad (6)$$

$$T_p = \eta \cdot \omega^2 \cdot d^3/4 \cdot [\pi \cdot l/h \cdot K - 1/4 \cdot \lambda_f (d+2h)] + T_0 \qquad (7)$$

where: T_a is the temperature on the shaft surface; T_p is the seal socket internal temperature; T_0 is the seal external environment temperature; K is the seal global thermal exchange coefficient; λ_f is the seal liquid thermal conductivity.

In figures 4 and 5 the temperature field variation on the *ox* and *oy* axis for a floating socket axial-cylindrical seal device is graphically presented, having the following characteristics: d = 20mm; h_0 = 0,02mm; l = 12mm; D = 40mm; ϖ = 150s^{-1}; Δp = $5 \cdot 10^5$ Pa; T_0 = 20^{0}C; sealed liquid, M30 oil.

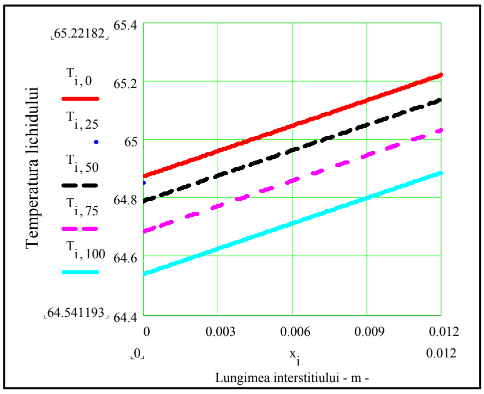

Figure 4 – The axial variation of the temperature field for a axial-cylindrical seal

Considering the figures it results that the determinative temperature increases linearly in the direction of the leakage flow-rate respectively decreases exponentially in radial direction, having the maximum point on the shaft surface. It is also noted that in both directions the temperature field has insignificant value variations (under 1^{0}C), fact that justifies the adopting, for the projecting calculus, of a unique temperature value for the entire liquid volume inside the clearance, the value of the determinative temperature.

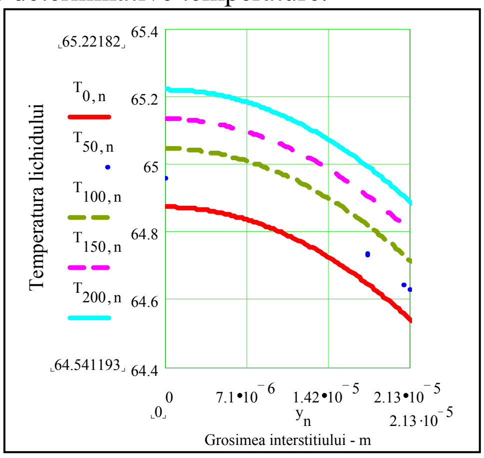

Figure 5 – The radial variation of the temperature field for an axial-cylindrical seal

For the same seal device, whose characteristics have been presented previously, figure 6 presents in comparison the results of the theoretical and experimental studies.

The theoretically calculated value of the sealed liquid temperature is greater than the one experimentally determined (the difference is yet very small, max. 2^0C) because when the determinative temperature calculus formula was established it was considered that all the heat caused by fluid friction is evacuated from the seal integrally through it's body; in reality a small part of this heat is evacuated through the leakage.

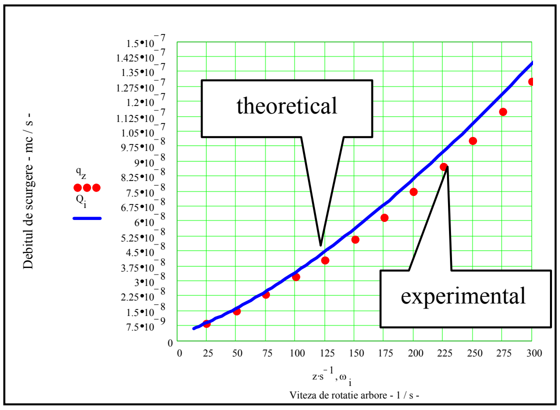

Figure 6 – $T_f = f(\omega)$

The theoretical studies also allowed the determination of the calculus relations for the seal socket superficial convection coefficients, for the global thermal exchange coefficient and for the seal socket external temperature.

Using equation (2) and the leakage flow-rate defining relation the following calculus formula was obtained for the leakage flow-rate:

$$Q = \pi\cdot[(\delta + a\cdot p_1)^4 - (\delta + a\cdot p_2)^4]\cdot \cdot[d + (\delta + a\cdot p_1)] \;/\; 48\cdot a\cdot l\cdot\eta \qquad (8)$$

Relation (8) points out the leakage flow-rate dependency of the constructive and dimensional seal parameters, respectively of the pressure fall on the clearance of the sealed liquid.

Because in the practice of non-contact seal devices projection the leakage flow-rate dependency of the thermal seal regime, respectively of it's thermal transparency matters a lot, using relations (3), (5) and (8), the following leakage flow-rate calculus expression was determined:

$$Q = \pi^2\cdot\omega^2\cdot d^3\cdot[(\delta + a\cdot p_1)^4 - (\delta + a\cdot p_2)^4]\cdot \cdot[d + (\delta + a\cdot p_1)] / 192\cdot a\cdot h\cdot K\cdot(T_f - T_0) \qquad (9)$$

Relation (9) emphasises the fact that the leakage flow-rate is inversely proportional to the power loss through friction inside the axial-cylindrical clearance, together with all the explicit and implicit consequences that follow from it.

Because functionally a maximum admissible determinative temperature, T_{fmax}, is most times imposed for a non-contact seal device, it explicitly results that the clearance width must be minimal (10), respectively the seal shaft rotation speed must be maximal (11); the corresponding leakage flow-rate values will also explicitly result.

$$h_{min} = \pi\cdot\eta_{Tmax}\cdot\omega^2\cdot d^3\cdot l \,/\, 4\cdot K_{Tmax}\cdot(T_{fmax} - T_0) \qquad (10)$$

$$\omega_{\max} = \sqrt{\frac{(T_{\max} - T_0)}{\eta_{T_{\max}}}\cdot\frac{1}{\frac{\pi\cdot d^2\cdot l}{4}}\cdot\frac{1}{\frac{d}{h_{T_{\max}}}}\cdot\frac{1}{\frac{1}{K_{T_{\max}}}}} \qquad (11)$$

Relations (10) and (11) constitute conditions for the limitation of the non-contact seal devices' applicability domain, although they explicitly show that the choosing of a higher value for the sealed liquid determinative temperature allows the utilisation of a smaller clearance width, thus of a smaller leakage flow-rate and of a higher seal shaft rotation speed.

The increase of the seal global heat exchange coefficient, *K*, leads directly to the increase of the seal efficiency through the increase of its capacity for heat evacuation out of the clearance, which implicitly leads to the possibility of decreasing the clearance width. The experimental researches pointed out certain analogies between the leakage flow-rates' values calculated with the established analytical relations, respectively the values measured during experiments, figure 7 (the graphs correspond to the same seal devices whose determinative temperature graphs were drawn in figure 6).

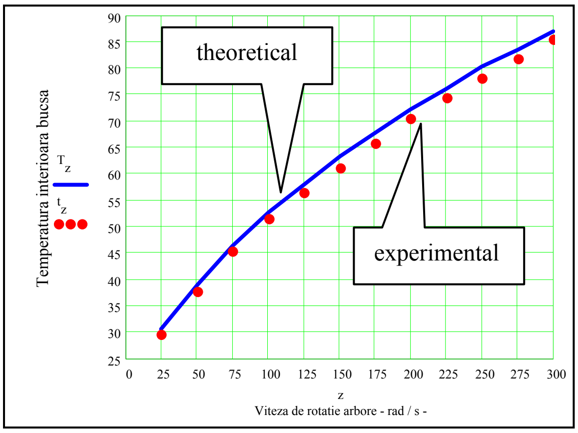

Figure 7 – $Q = f(\omega)$

The cause that led to the experimental recording of smaller leakage flow-rates was that the actual value of the sealed liquid determinative temperature is lower than the theoretical one.

4. CONCLUSIONS

- The effectiveness of non-contact seal devices with axial-cylindrical clearances is determined by the size of the leakage flow-rates and by the possibility of their diminution; in normal conditions the leakage flow-rates' annulment is impossible.
- The developed theories and the determined analytical relations during the theoretical studies were validated by the experimental studies' results.
- The power loss through friction proportional to the clearance form and dimensions, to the square of the seal shaft rotation speed and to the sealed liquid viscosity.
- The temperature field distribution inside the seal clearances has demonstrated that the sealed liquid temperature has insignificant variations inside the volume of the liquid inside the clearance and so it is possible to take in consideration a unique, average temperature, called determinative temperature, that is proportional to the power loss through friction, to the global seal thermal exchange coefficient and to the seal's external environment temperature value.
- Increasing the global seal thermal exchange coefficient (the determinative temperature rises and the viscosity decreases) can also diminish the leakage flow-rates.
- Imposing certain minimum values for the leakage flow-rates implies restricting conditions for the shaft rotation speed or for the sealed liquid determinative temperature.

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